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by Robert G. Dorsch

Lewis Research Center
Cleveland, Ohio

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ABSTRACT

E-3928

The frequency response of a pump-fed single-tube heat exchanger boiler employing Freon 113 as the boiling fluid and hot water as the heating fluid was determined experimentally. An electro-hydraulic servo-valve was used to impose quasi-sinusoidal perturbations on the mean flow into the boiler. Frequencies between 0.04 and 4 Hz were employed. Inlet pressure and flow perturbations were measured with dynamic pickups. A transfer function analyzer (tuned only to the fundamental imposed frequency) was used to obtain the amplitude and phase of each pressure and flow response.

From this data the boiler inlet impedance as a function of frequency was determined for a variety of steady state operating conditions. Primary steady state variables were heat flux and exit quality. Data were obtained with two boiler tube configurations: a hollow tube, and a tube with a spring-plug insert to swirl and turbulate the flow.

A brief summary of the experimental data is presented. The boiler inlet impedance plots indicate that feed-system coupled instabilities will occur at many of the steady state operating conditions if sufficient feed-system stabilization is not provided. The impedance data provide a measure of the amount of feed system stabilization needed. The data also can be used to predict the frequency at which a given system will go into natural oscillation.

INTRODUCTION

Two phase stability problems are frequently encountered in the development of systems in which liquid flows through heated passages (ref. 1). The Rankine cycle turboelectric power plant with a reactor heat source is typical of this class of system.

A key component of the Rankine cycle system is the boiler or vaporizer. For space vehicle applications the system must operate in a zero or low gravity environment and will employ low vapor pressure alkali metals as the working fluid. In contrast to earth based systems the boil-

ing mode is therefore likely to be forced flow at low pressure rather than natural circulation at high pressure. Stability problems are increased with low boiler pressures because large increases in specific volume occur as the liquid vaporizes.

In low pressure forced flow boilers an important type of low frequency instability results from the interaction between the flow supplied by the feed system and the pressure determined by the boiler and its exit hardware. This type of instability has frequently been encountered (at higher frequencies) in liquid rocket engines where it is known as chugging. The coupling mechanism for feed-system combustion-chamber interaction in a monopropellant rocket (refs. 2 and 3) is very similar to that for a boiler and its feed system (ref. 4).

In order to describe this interaction analytically the dynamic characteristics of the feed-system, boiler, and exit hardware (system components downstream of the boiler) must be known. Since little is known about boiler dynamic characteristics an experimental and analytical study was initiated at the NASA Lewis Research Center. The objective is to obtain verified analytical models for dynamic representation of forced flow boilers in terms of measurable steady state parameters.

In the first phase of the study experimental data were obtained with self-induced flow and pressure oscillations in or near the onset of an unstable operating condition. These data (refs. 4, 5, and 6) were found to be difficult to interpret in dynamic terms. Recent emphasis has therefore been on forced oscillation experiments at stable boiler operating conditions. In these experiments frequency response techniques used successfully in studies of the dynamics of hydraulic and liquid rocket propellant systems (refs. 7 and 8) were employed to measure the dynamic impedance of the boiler inlet (complex ratio of inlet pressure perturbation to inlet flow perturbation). The use of these techniques is based on the hypothesis that for small amplitude disturbances a boiler can be treated as if it were a linear element.

This paper gives a brief summary of the Lewis Research Center single tube boiler frequency response studies which have been completed to date (refs. 9 and 10) and discusses the implications of the results on the stability of forced-fed boiler systems.

APPARATUS

The test facility is shown schematically in figure 1. The boiler test section was a shell and tube heat exchanger with hot water flowing counter-currently in the annulus between the tube and shell. Freon 113 (trichlorotrifluoroethane) flowed within the tube. A gear pump forced the liquid

Freon through the vertically mounted boiler tube where it was vaporized. The boiler tube exit was located in a plenum chamber connected by a low pressure drop line to a large water cooled condenser which was vented to the atmosphere to provide constant pressure. An accumulator downstream of the pump acted as a quasi-constant pressure supply for the electro-hydraulic controlled valve used to impose flow oscillations on the mean flow into the boiler. The valve open area was varied sinusoidally about a mean value in response to a command oscillator signal.

Two different test sections (fig. 2) were employed in the study. The first (fig. 2(a)) had a hollow copper boiler tube with an I.D. of 8.0 mm and a wall thickness of 0.76 mm. The heated length was 91.4 cm. The second boiler test section (fig. 2(b)) had a stainless steel tube with an I.D. of 10.92 mm and a wall thickness of 0.89 mm. The heated length was 81.3 cm. This tube had a spirally wound spring in the two phase region and an inlet plug insert in the subcooled liquid region as shown in figure 2(b). The inlet plug was a rod which made contact with the spring so that the entering subcooled liquid flowed through a spiraled annular channel.

Steady state flow rates, pressures, and temperatures were measured at the locations given in figures 1 and 2 and Table I. Freon perturbation pressures and flows at the boiler inlet were measured with a quartz crystal pressure transducer and a fast response turbine flowmeter, respectively. The flow and pressure perturbation signals from the dynamic pickups were processed with a transfer function analyzer. This device made a Fourier analysis of the perturbation signals. This analysis isolated the fundamental sinusoidal component of each signal at the test frequency and computed its amplitude and phase relative to the command oscillator signal.

PROCEDURE

Dynamic data were obtained with each test section at several heat flux levels and for a range of exit vapor qualities. The water flow rate was held at approximately the same value (0.088 kg/sec) for all runs. The inlet water temperature was adjusted to a selected level and held constant over a range of Freon mean flow rates. At a given inlet water temperature the Freon mean flow rate was varied in order to obtain different exit qualities. The inlet temperature of the Freon was held constant during each frequency response run.

Frequency response data were taken at the selected steady state conditions over a frequency range from 0.04 to 4 Hz. The amplitude of the servo-valve area variations was maintained constant over the entire range of frequencies for each run. In addition, the amplitude of the area oscillation was kept small compared to the mean open area of the valve in order to reduce nonlinear effects.

At each frequency the amplitude and phase angle (relative to the command sine wave signal) of the boiler inlet pressure and flow sinusoids were obtained from the transfer function analyzer. From this data the complex ratio of inlet pressure perturbation to flow perturbation was obtained. Inasmuch as the pressure in the plenum tank at the boiler exit was maintained constant, this complex ratio is the inlet impedance of the boiler.

FREQUENCY RESPONSE RESULTS

Steady-State Operating Conditions

Frequency response data were obtained with each test section at several heating fluid temperature levels and are reported in detail in references 9 and 10. For this paper, representative data were selected from runs which (with one exception) had a nominal water inlet temperature of 372° K. The steady-state data for the dynamic runs selected are given in Table II. The steady-state pressure drop as a function of Freon mass flow rate is shown for the two test sections in figure 3. Exit qualities (in percent) are given adjacent to the data points. The operating points for the dynamic runs of Table II are shown as solid symbols.

For the conditions of Table II it was found that crosscoupling effects between the boiling and heating fluid sides of the heat exchanger boiler were small. That is, during the dynamic runs the perturbations in water outlet temperature were found to be negligible.

Boiler Inlet Impedance

The inlet impedance of the 8.0 mm hollow tube boiler is given in polar coordinate form in figure 4. The real and imaginary axes of the corresponding complex plane are also indicated in the figure. Data for three different exit qualities are shown. The amplitude of the impedance is the distance from the origin to the locus curve. Note that the amplitude scales differ considerably for parts (a), (b), and (c) of the figure because of the large change in amplitude with change in exit quality. With increasing frequency the amplitude vector rotates in a clockwise direction. The angle measured clockwise from the positive real axis is the phase lag of the pressure perturbation with respect to the flow perturbation. This phase lag generally increases with frequency as can be seen from figure 4.

The curves of figure 4 show that the real part of the dynamic impedance is negative (phase angles between -90 and -270 degrees) over a considerable range of frequencies at each steady state operating condition. The existence of negative boiler impedance implies that there is the potential for instability as will be discussed later.

The data of figure 4(a) were taken at a low exit quality (13 percent) operating point in the negative resistance (negative slope) region of the steady state pressure drop against flow curve (fig. 3). As would be predicted from the negative resistance operating point the boiler dynamic impedance of figure 4(a) starts with a -180 degree phase angle at low frequency. The magnitude of the boiler impedance at -180 degrees is small because the steady-state pressure drop curve (fig. 3) has only a small negative slope at this operating point. As the frequency increases the magnitude vector rotates in the clockwise direction so that the real part of the impedance becomes positive at frequencies above 0.6 Hz. Thus, the boiler becomes inherently stable above 0.6 Hz.

The data of figures 4(b) and (c) were taken at operating points in the positive slope region of the steady-state pressure drop against flow curve of figure 3. Predictions based on these steady-state operating points indicate that the real part of the dynamic impedance (resistance) should be positive at very low frequencies. Figures 4(b) and (c) confirm these predictions. It should also be noted that the magnitude of the impedance at low frequencies tends to increase with quality. This results from an increase in boiler resistance as the vapor void fraction increases.

The curves of figure 4 show that the impedance phase angle continually becomes more negative (the lag increases) with frequency. This is typical of response in which dead time makes an important contribution to the dynamics. The shape of figure 4(c), for example, is typical of the impedance locus of a rocket engine combustion chamber where the dead time is the amount of time required for the fuel to vaporize and react. The dead time in the boiler is believed to be the time required for a particle to travel from the boiler inlet to the end of the subcooled liquid region (ref. 4). That is, the subcooled dead time is obtained by dividing the subcooled length (given in Table II) by the liquid velocity. The effect of dead time can be seen by comparing the response of run 2 with that of run 3. Run 2 (fig. 4(b)) and run 3 (fig. 4(c)) have calculated subcooled dead times of 0.79 and 0.27 seconds, respectively. In agreement with these values the magnitude vector rotates with frequency more than twice as fast in figure 4(b) as in figure 4(c). For example, the impedance loci crosses the negative real axis (-180 degree phase angle) at a frequency of 0.43 Hz in run 2 and at a frequency of 1.05 Hz in run 3. The phase angle is, of course, also affected by other boiler parameters.

The boiler inlet impedance for the 10.92 mm I. D. boiler with swirl-turbulator inserts is given in figure 5. Data are shown for exit qualities of 20 , 75 , and 99 percent and for 100 percent with 5.6° K superheat. In general, the data of figure 5 for the boiler with inserts have similar impedance loci to those for the hollow tube boiler (fig. 4). Figure 5(a) is an exception, however, in that at this low exit quality (20 percent) the boiler

impedance has a positive real part over the entire frequency range. At higher exit qualities (figs. 5(b) and (c)) the impedance has a negative real part over some portion of the frequency range as was found for the hollow tube boiler. Achievement of vapor superheat at the boiler exit (fig. 5(d)) resulted in frequency response data grossly similar to that obtained at 99 percent quality. For this run the water inlet temperature was raised to 383° K as superheat was not obtainable at the 372° K temperature level.

STABILITY IMPLICATIONS

Feed System Coupled Instability

A negative real part of the boiler inlet dynamic impedance (i.e., negative resistance) exists over a significant part of the frequency range of all except one of the polar data plots shown. This implies that feed-system-coupled hydrodynamic instabilities are possible. This is particularly true at the higher exit qualities where the larger values of negative impedance were measured.

Resistive feed system. - Consider the boiler used in this study (with constant pressure at the exit) to be supplied by a hypothetical feed system which is entirely resistive. This would correspond physically, for example, to the boiler being fed through a resistive orifice in a constant pressure tank. For a given boiler test section and steady state operating condition the magnitude of the boiler impedance at -180 degrees (figs. 4 and 5) is then a measure of the amount of feed-system hydraulic resistance needed for neutral stability. Feed system dynamic resistances larger than this value would permit stable operation of the boiler at the selected operating condition. Resistances equal to or less than this value would result in system flow and pressure oscillations. The frequency of the natural oscillations would correspond to the perturbation frequency for which the boiler dynamic impedance had the -180 degree phase angle. More than one system oscillation frequency is possible at some steady state conditions. For the boiler without inserts at 63 percent exit quality (fig. 4(c)), instability frequencies of 1 and 4 Hz are possible. It is also seen that greater feed system resistance would be required to prevent an instability at 1 Hz than at 4 Hz.

Complex feed system. - Forced flow boiler feed systems, including the one used in this study, are usually complex devices that contain not only resistance but hydraulic inertance and compliance as well. These latter parameters are a function of frequency. Therefore, the feed system dynamic impedance is a function of frequency as well as steady state operating conditions. To illustrate this point, typical impedance loci for the boiler and feed system of this study are shown in figure 6. The solid feed system curve in figure 6 represents the impedance as a function

of frequency at a typical high impedance steady state condition for a frequency response run. The locus shown is for the servo-valve at the half-open position and stationary. At this operating condition the feed system provides sufficient boiler stabilization so that the system will not go into natural oscillation while obtaining the forced (by oscillating the valve about the mean open position) response curve shown for the boiler. The dashed curve of figure 6 represents the feed system response when the servo-valve is adjusted to a more open position while maintaining the same supply flow to the boiler. Because of the drop in servo-valve dynamic resistance the feed system provides less stabilization. It should be noted that a change in the feed system dynamic impedance has no effect on the boiler curve so long as the steady state flow (and boiler conditions) remain the same.

In this complex system (constant pressure at the boiler exit) a hydraulic instability will occur if at some frequency the feed system dynamic impedance is equal to (or less than) the boiler dynamic impedance and 180 degrees out of phase with the boiler impedance. This condition (for neutral stability) is illustrated in figure 7 for two different steady state operating conditions. For simplicity, only boiler impedances between 0.04 and 0.8 Hz and feed system impedances between 0.04 and 4 Hz are shown. The feed system impedances shown in figures 7(a) and (b) are for highly open servo-valve positions. Neutral stability conditions for the boiler - feed-system exist at 0.44 and 0.475 Hz, respectively.

Experimental Check

The validity of the overall approach used in this study was checked experimentally as follows. The frequency response runs shown in figures 4 and 5 were taken at stable steady-state (or mean) operating conditions. At the completion of each run the servo-valve (no longer oscillating) was adjusted to a more open position. The pump rotation speed was then adjusted to provide the same Freon flow rate into the boiler as was used during the frequency response run. This sequence was repeated using small steps in valve position. Each step change decreased the dynamic impedance of the feed system by a small amount. Pressure and flow signals were continuously recorded during the sequence. The system went into natural oscillation (flow and pressure) for the two cases shown in figure 7. For both these cases the oscillations occurred when the servo-valve was almost fully open. The resulting pressure and flow traces from the oscillograph are shown in figure 8. The oscillations shown in figure 8 have already grown to the point where their amplitude is limited by system nonlinearities. Figure 8(a) shows that for the 99 percent quality run the natural oscillation frequency is 0.44 Hz and the phase difference between the flow and pressure is approximately 160 degrees. For the superheat condition (fig. 8(b)) these values are 0.475 Hz and 170 degrees.

The frequency and phase difference values obtained from the natural oscillation traces for the two runs of figure 8 are very close to the neutral stability frequency and phase difference predicted by the dynamic data (fig. 7). Natural oscillations did not occur upon opening the servo-valve at the conclusion of many of the runs because the magnitude of the boiler negative impedance was so small that the feed system impedance was larger even with the valve fully open. That is the pipe friction pressure drop alone provided sufficient resistance to stabilize the boiler. These results indicate that frequency response data can be used to predict feed-system-coupled boiler instabilities.

Effect of Boiler Inserts

In order to obtain high (near 100 percent) quality vapor from a single tube boiler some type of swirl-turbulator insert is usually required to prevent transition to film boiling. The frequency response data obtained with the test section having the spring-plug insert indicate that at the higher exit qualities (figs. 5(b) through (d)) the insert used had no major effect on the dynamic response. At low exit qualities (fig. 5(a)) there was a strong stabilizing effect demonstrated by the real part of the boiler inlet impedance being positive over the entire frequency range. The boiler would therefore be stable with any passive feed system. The plug and spring insert appears to increase stability at low exit qualities because the liquid resistance and inertance in the channel region of the plug is large compared to the dynamic resistance of the two phase region.

CONCLUDING REMARKS

The results of this study indicate that measured boiler response to forced flow oscillations at a given stable steady-state (mean) operating condition can be used to predict the occurrence of natural flow and pressure oscillations in a system. That is, if the dynamic response of a particular boiler and its feed supply is known then the stability of the system can be determined. If, as is often the case, a boiler does not have constant pressure at the exit, then boiler inlet to exit transfer functions as well as the dynamic impedance of the exit hardware must be measured or calculated. Under some circumstances it may be convenient to measure at the boiler inlet the combined impedance of the boiler and downstream hardware by frequency response techniques.

Thus boiler frequency response data should prove to be useful for making stability calculations for a specific system at its operating condition. The primary use of this type of data, however, is to identify the controlling dynamic parameters of the boiler and to evaluate and verify analytical models as they are being developed.

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TABLE I. - LOCATION OF WATER TEMPERATURE STATIONS

Station number	Distance from datum line			
	Hollow tube boiler (fig. 2(a))		Insert boiler (fig. 2(b))	
	cm	in.	cm	in.
1	0.8	5/16	1.9	3/4
2	10.2	4	5.1	2
3	20.3	8	15.2	6
4	30.5	12	25.4	10
5	40.6	16	35.6	14
6	60.9	24	45.7	18
7	71.1	28	55.9	22
8	81.3	32	66.0	26
9	90.6	35 $\frac{11}{16}$	76.2	30
10	----	--	79.4	31 $\frac{1}{4}$

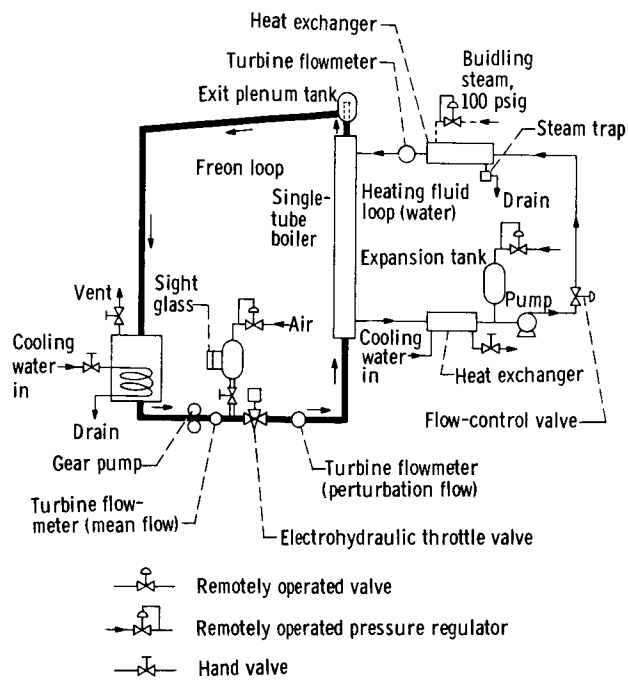
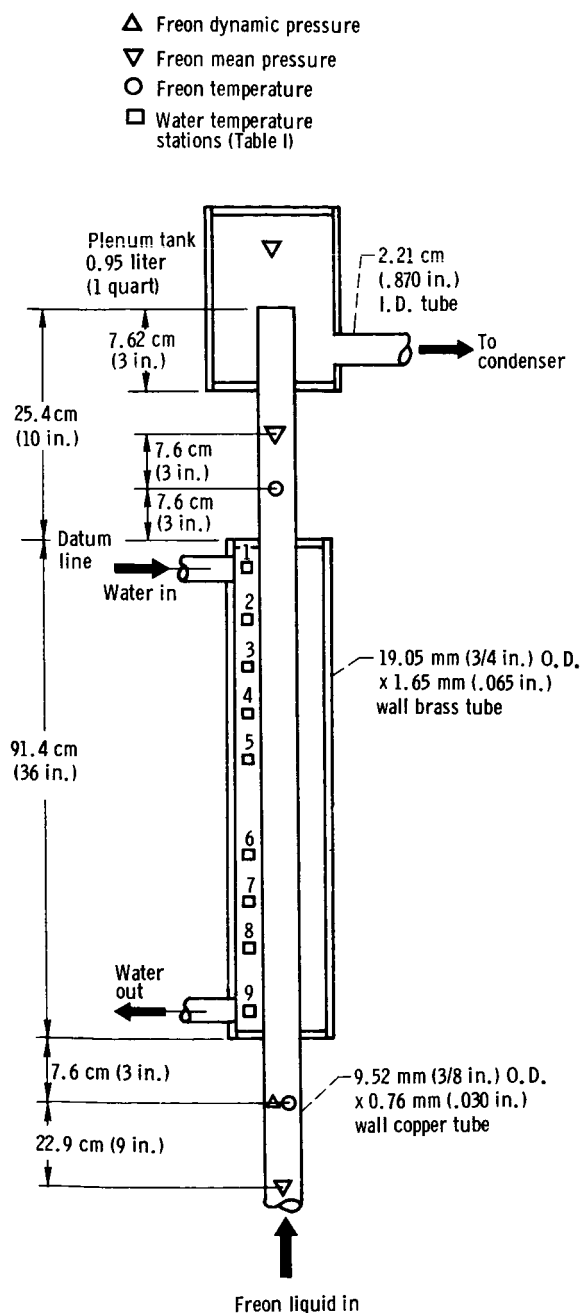
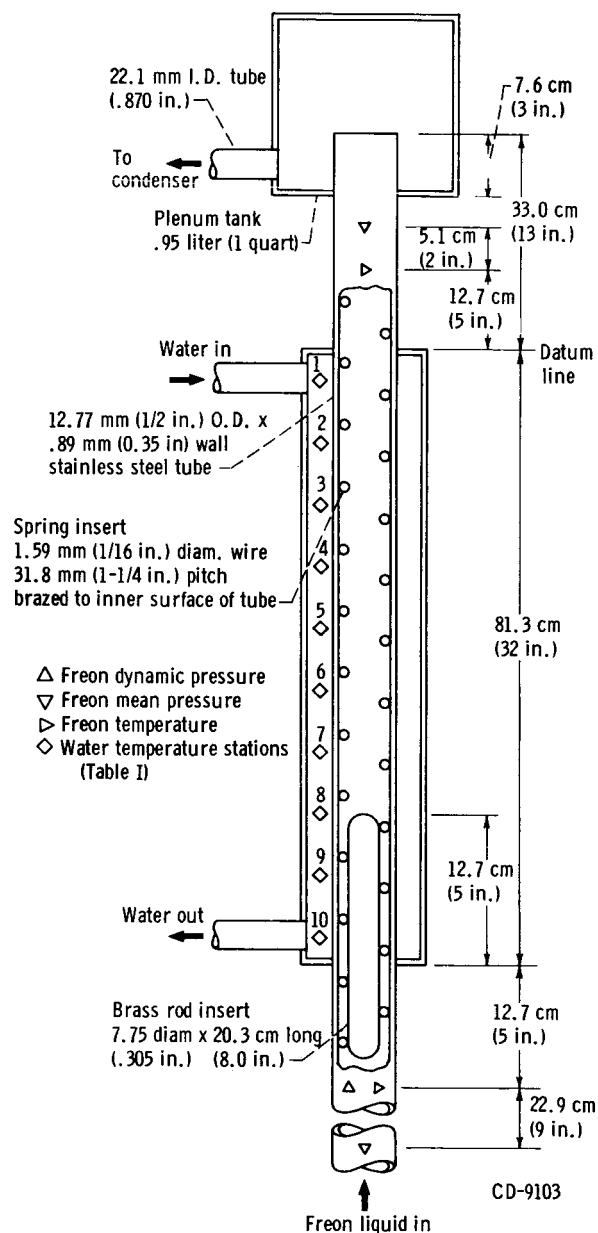


Figure 1. - Boiling dynamics facility.



(a) Hollow tube configuration.

Figure 2. - Single-tube boiler test sections.



(b) Tube with spiral spring and plug inserts.

Figure 2. - Continued. Single-tube boiler test sections.

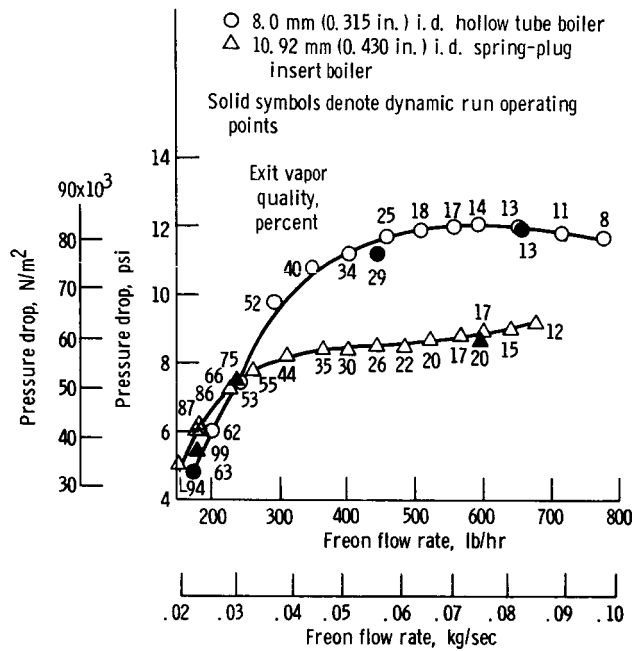
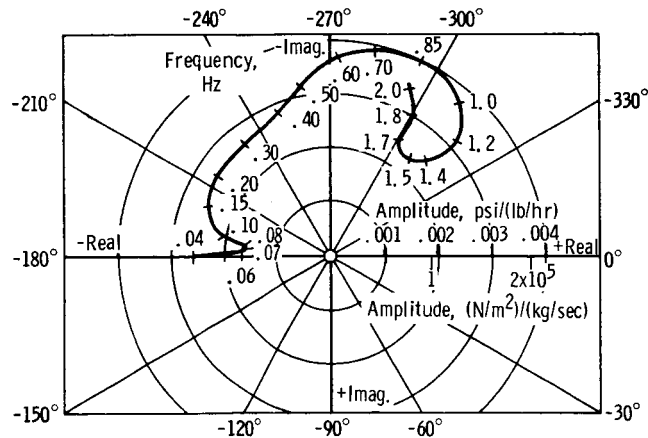
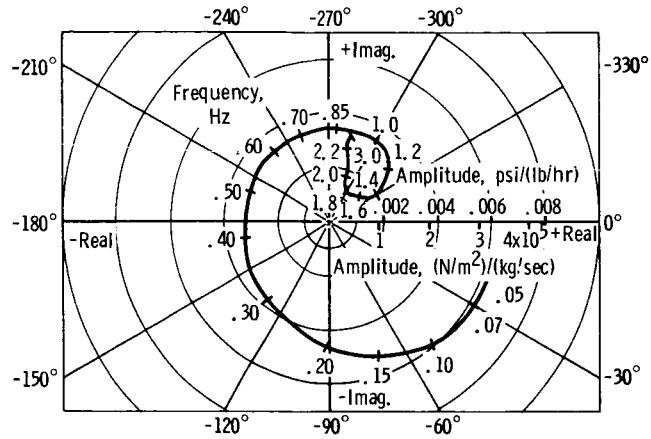


Figure 3. - Steady-state (or mean) boiler pressure drop as a function of mass flow rate and exit quality. Water inlet temperature, 372° K (210° F).



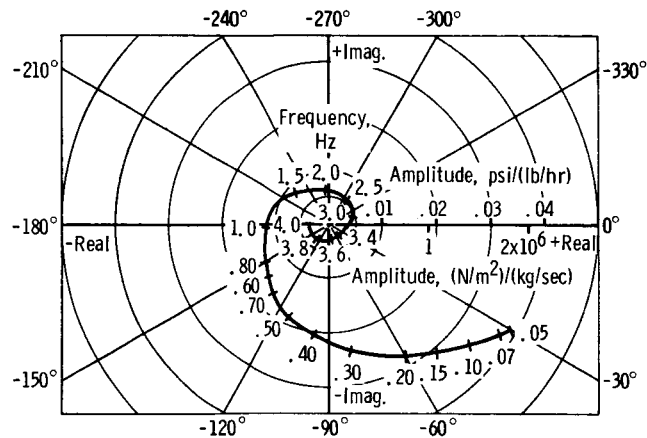
(a) Run 1, table II. Exit vapor quality, 13 percent.

Figure 4. - Inlet impedance of hollow-tube boiler.



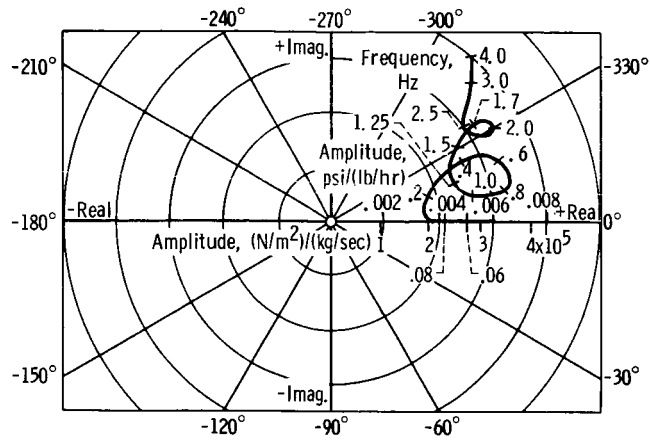
(b) Run 2, table II. Exit vapor quality, 29 percent.

Figure 4. - Continued.



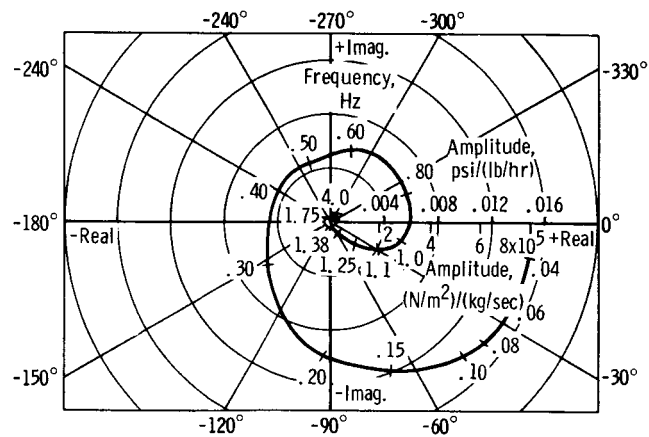
(c) Run 3, table II. Exit vapor quality, 63 percent.

Figure 4. - Concluded.



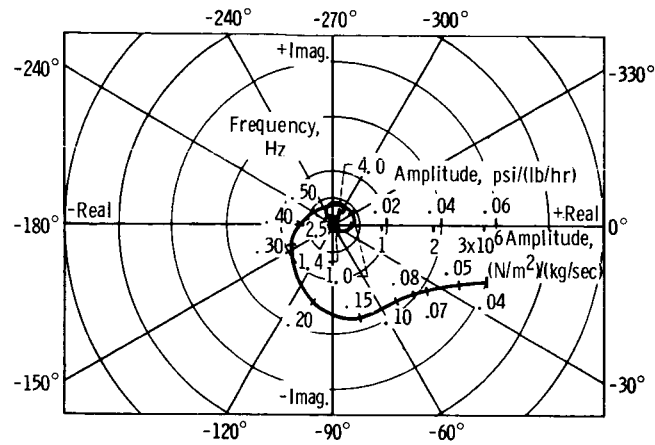
(a) Run 4, table II. Exit vapor quality, 20 percent.

Figure 5. - Inlet impedance of boiler with inserts.



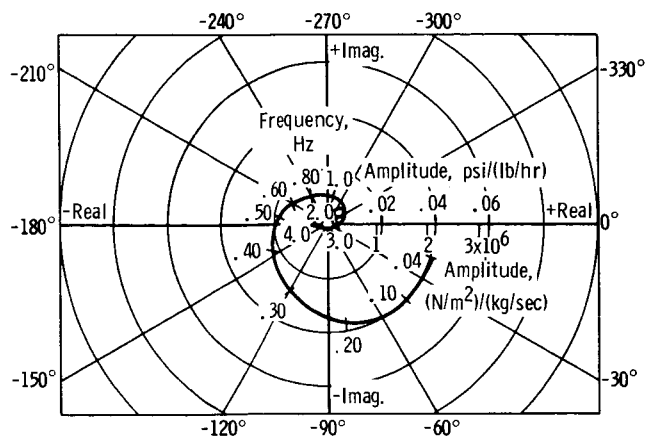
(b) Run 5, table II. Exit vapor quality, 75 percent.

Figure 5. - Continued.



(c) Run 6, table II. Exit vapor quality, 99 percent.

Figure 5. - Continued.



(d) Exit vapor quality 100 percent with 5.6° K (10° F) superheat.
Run 7, table II.

Figure 5. - Concluded.

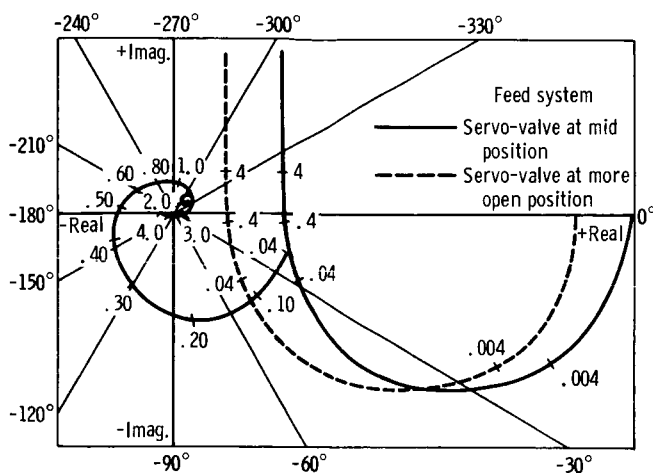
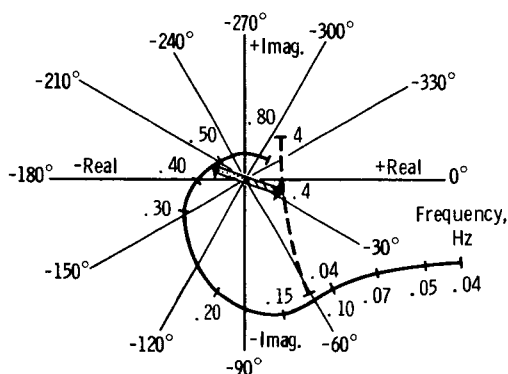
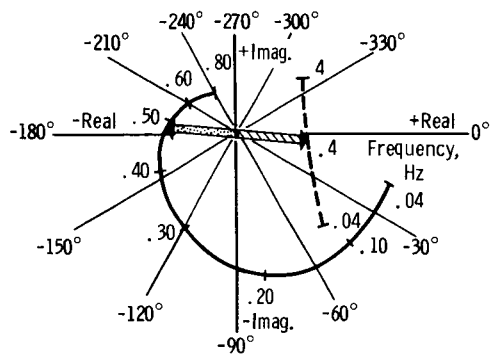


Figure 6. - Typical impedance plots for boiler inlet and feed system at boiler inlet.

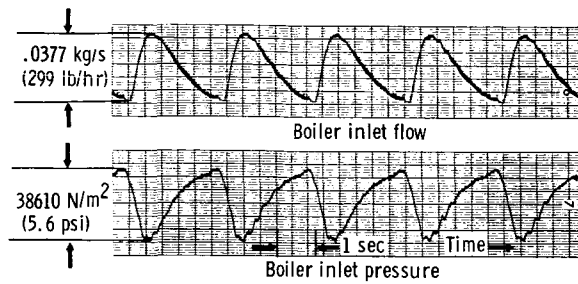


(a) Run 6, table II. Exit quality 99 percent.

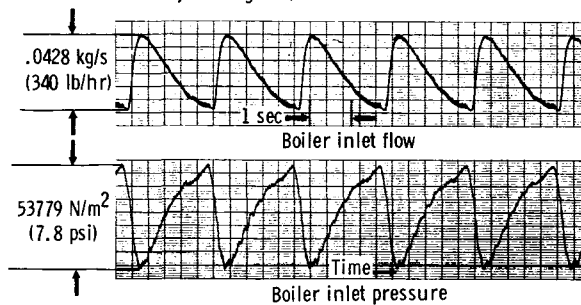


(b) Run 7, table II. Exit quality 100 percent, superheat 5.6° K.

Figure 7. - Neutral stability condition for boiler and feed system.



(a) Boiler conditions same as run 6, Table II. Exit quality 99 percent. Natural oscillation frequency, 0.44 Hz. Phase difference between pressure and flow, 160 degrees.



(b) Boiler conditions same as run 7, Table II. Exit quality 100 percent with 5.6 deg. K superheat. Natural oscillation frequency 0.475 Hz. Phase difference between pressure and flow, 170 degrees.

Figure 8. - Oscilloscope traces of natural pressure and flow oscillations. The instabilities shown have reached their respective amplitude limits for the existing boiler and feed system conditions.